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Design, Fatigue, and Strength Analysis of a Tie Rod Hydraulic Cylinder Bolt

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ABSTRACT

This report illustrates strength and fatigue analysis completed on a tie rod hydraulic cylinder bolt from a Lion TX 2500 tie rod hydraulic cylinder. The goal was to: solve for total clamping force, damping force per bolt at 90% proof load strength, amount of force per bolt, find the required number of bolts to hold the hydraulic cylinder together, solve for bolt spacing requirements, estimate a safety factor based on proof load strength, nut shear strength, required nut thickness, the least number of threads that must be engaged, estimate R.R. Moore Bending Fatigue Strengths, estimate the maximum value of P that will not cause eventual fatigue failure, and estimate a safety factor with respect to eventual bolt fatigue failure. Calculations show that the hydraulic cylinder must be built using four 0.5 in bolts with a minimum of nine threads engaged. The hydraulic cylinder is capable of supporting a constant load of 22,704 lb without eventual fatigue failure. Safety factors were calculated based on proof load strength with respect to eventual bolt fatigue failure to be 4.11 and 2, respectively. This analysis validated the manufacturer's design and specifications of the hydraulic cylinder.

INTRODUCTION

A hydraulic cylinder is a mechanical actuator that can be used to cause a unidirectional force through a unidirectional stroke. A hydraulic cylinder is used to transfer a force applied at one point to another point using an incompressible fluid. Hydraulic cylinders are often powered from pressurized hydraulic fluid or oil. The hydraulic fluid is pressurized by a hydraulic pump which forces fluid in to the cylinder during the extension stroke. The fluid pressure on the piston is equivalent to the pressure on the piston rod. During the retraction stroke, the fluid flows back to the hydraulic reservoir.

A hydraulic cylinder consists of a cylinder barrel, piston, and piston rod. The piston contains sliding rings and seals and is closed on the top by a cylinder head or gland. The cylinder is closed on the bottom by a cylinder bottom or cap.

The cylinder analyzed uses four high strength threaded rods to hold the end caps on. Unlike some hydraulic cylinders, which are welded together, a tie rod hydraulic cylinder uses four rods with threaded ends to hold the end caps to the main cylinder wall - allowing the cylinder to be easily taken apart for maintenance and replacing or refurbishing individual components.

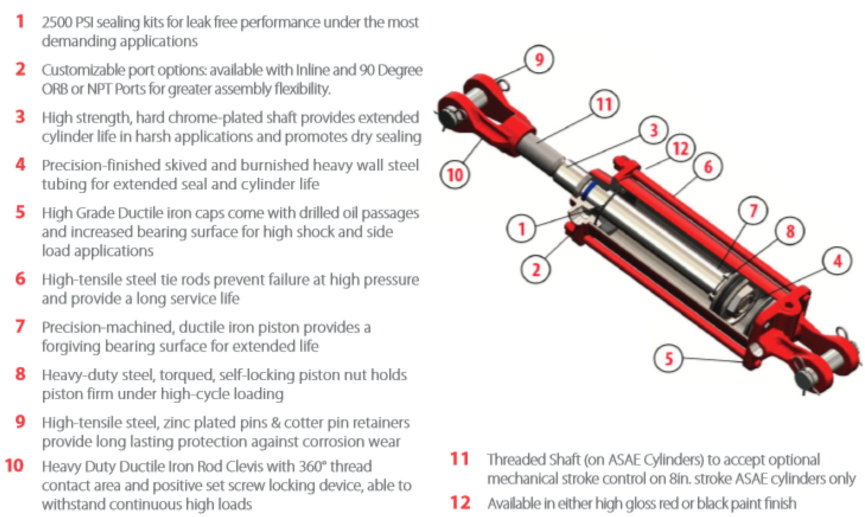

Figure 1. The TX 2500 Tie Rod Hydraulic Cylinder [2]

The tie rod hydraulic cylinder bolts were chosen to be analyzed because they are a point where the system fails. The bolt itself may not be able to hold the pressure supplied to the cylinder, the bolt threads could fail, or the nut could fail to resist the maximum load. When under pressure, the bolts must keep the end caps on and the assembly together. The bolts undergo tensional loading, with the maximum pressure being the maximum output of the hydraulic pump.

Tie rod hydraulic cylinder bolts were analyzed using knowledge and concepts learned from Design of Machine Element’s course material. Factory specifications from a Lion TX 2500 tie rod hydraulic cylinder were used to complete a design, strength, and fatigue analysis; to determine the number of bolts needed to hold the cylinder together, the bolt size required to resist eventual fatigue failure, nut size, required number of engaged threads, safety factors, and the maximum possible constant load.

The hydraulic cylinder bolts and the entire hydraulic cylinder were drawn using Solid Works CAD software. An actual TX 2500 hydraulic cylinder was used to record the desired measurements to complete a 3D model in Solid Works. The Solid Works program was also used to create 2D drawings from the 3D model.

METHODS

Phase I Procedure:

- Acquire TX 2500 Tie Rod Hydraulic Cylinder Specifications
- Disassemble TX 2500 Hydraulic Cylinder
- Record Proper Measurements of Necessary Cylinder Components
- Complete a Sketch of Components
- Construct 3D Model Using Solid Works
- Use 3D Model to Construct 2D Drawings

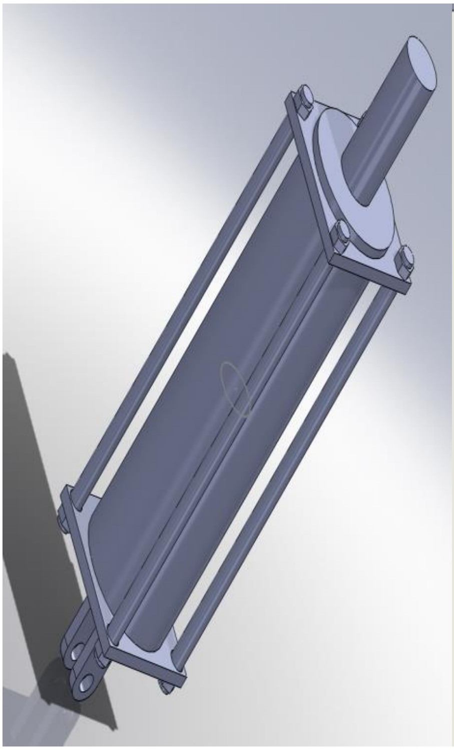

Figure 2. 3D TX 2500 [2]

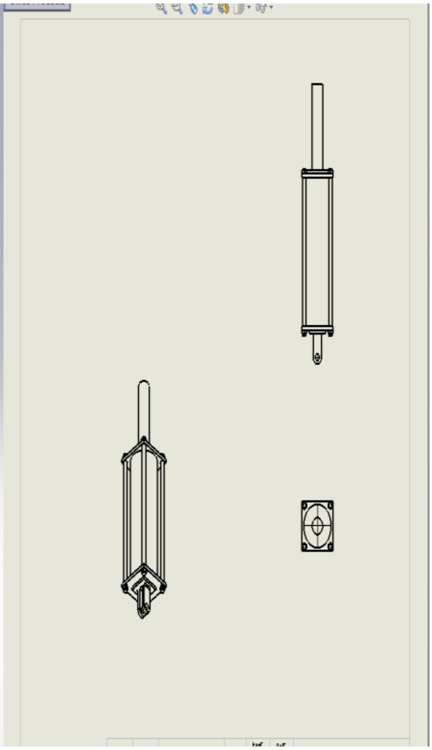

Figure 3. 2D TX 2500 Drawings [2]

Figure 4. TX 2500 Bolt [2]

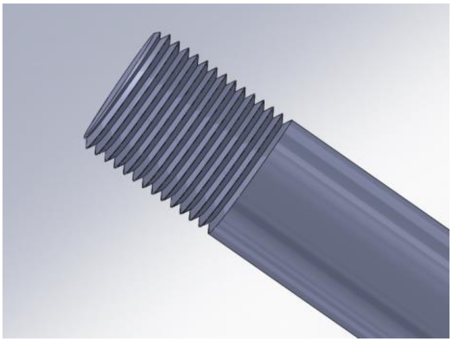

Figure 5. TX 2500 Bolt Threads [2]

Phase II Procedure:

- The Following Assumptions were used:

  1. Uniform Internal Pressure
  2. Bolt Loading is Static
  3. Load is Equally Distributed among Bolts
  4. Grade 8 Bolts
  5. Coarse Threads
  6. Fine Ground Steel/Polished
  7. Nut is Made of SAE Grade 2 Steel
  8. $F_i = 0.9(A_t)(S_p)$
  9. $K_c = 4K_b$

- Solve for Total Clamping Force
- Solve for Damping Force per Bolt @ 90% S_p
- Solve for Force/Bolt
- Find Required Number of Bolts
- Solve For Spacing Requirements
- Estimate Safety Factor Based On Proof Load Strength
- Solve For Nut Shear Strength
- Solve For Nut Thickness
- Solve For Least Number of Threads That Must Be Engaged
- Estimate R.R. Moore Bending Fatigue Strengths
- Estimate Maximum Value of P That Will Not Cause Eventual Fatigue Failure
- Estimate Safety Factor With Respect to Eventual Bolt Fatigue Failure

Bolt Requirement

The required number of bolts for a pressure vessel with a gasketed end plate can be found using Equations 1, 2, and 3. The internal pressure is assumed to be uniform and the bolt loading can be considered static. The bolts used are 0.5 in; SAE Grade 8 bolts with coarse threads.

Equation 1. $$Total\ Clamping\ Force = (pressure)\left(\frac{\pi}{4}\right)\left(D_o^2 - D_i^2\right)$$

Nomenclature:

$D_o = outside\ diameter$

$D_i = inside\ diameter$

Damping Force

Damping force acts to counter the current motion of the object and acts to damp out any motion in the object. Damping force per bolt is typically 90% of the proof load strength. The following equation can be used to find the damping force per bolt at 90% S_p.

Equation 2. $$Damping\ Force\ Per\ Bolt\ @\ 90\%\ \ S_p = (A_t)(0.9)(S_p)$$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_p = Proof\ Load\ Strength$

Equation 3. $$\frac{Force}{Bolt} = \frac{Total\ Clamping\ Force}{Damping\ Force\ Per\ Bolt}$$

Equation 4 and 5 were used to find the number of bolts needed to meet the spacing requirement of having a ratio of bolt circumference to the number of bolts that does not exceed 10, nor be less than 5.

Equation 4. $$Spacing = \frac{(Distance\ Between\ Nonadjacent\ Bolts)(\pi)}{\#\ of\ Bolts}$$

Equation 5. $$Ratio = \frac{Spacing}{Bolt\ Diameter}$$

Safety Factor

A safety factor is a term that describes the structural capacity of a system beyond the expected loads or actual loads; i.e., the amount the system is stronger than it needs to be to handle an intended load. Safety factors can be calculated using the following equation.

Equation 6. $SF = \frac{Design\ Overload}{Normal\ Load} = \frac{(S_p)(A_t)}{Normal\ Load}$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_p = Proof\ Load\ Strength$

Thread Engagement Requirement

The nut is made of a softer material than the bolt so that the highly loaded first thread will deflect (either elastically or plastically), thereby transferring more of the load to other threads. The nut threads are manufactured with a slightly greater pitch than that of the bolt threads so that the two pitches are theoretically equal after the load is applied. Equations 7, 8, 9, and 10 can be used to provide a balance between bolt tensile strength and thread stripping strength. If the bolt size and nut thickness are known, the required number of engaged threads can be calculated.

Equation 7. $Tensile\ Force = A_t S_y$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_y = Yield\ Strength$

Equation 8. $S_{ys} = 0.58 S_y$

Nomenclature:

$S_{ys} = Shear\ Yield\ Strength$

$S_y = Yield\ Strength$

Equation 9. $Nut\ Shear\ Force = (\pi)(d)(0.75t)(S_{ys})$

Nomenclature:

$S_{ys} = Shear\ Yield\ Strength$

$S_y = Yield\ Strength$

$t = thickness$

Equation 10. $\#\ of\ Thread\ \ Engagement\ = (Thickness)(Threads\ Per\ in)$

10^6, 10^3, and 2×10^5 Cycle Strength for Fatigue Life (Axial Loading)

Axial loading subjects the entire cross section to the maximum stress. Fatigue failure results from repeated plastic deformation. Without repeated plastic yielding, fatigue failures cannot occur. The Moore [1] fatigue tests are used to determine the fatigue strength characteristics of materials under a standardized and highly restricted set of conditions.

$10^6\ Life\ Cycles$

Equation 11. $S_n = S_n^{'} C_L C_G C_s C_T C_R$

Nomenclature:

$S_n = 10^6\ Cycle\ Strength\ For\ Fatigue\ Life$

$S_n^{'} = 0.5S_u = 10^6\ Cycle\ Strength\ For\ Axial\ Loads$

$S_u = Tensile\ Strength$

$C_L = Load\ Factor$

$C_G = Gradient\ Factor$

$C_s = Surface\ Factor$

$C_T = Temperature\ Factor$

$C_R = Reliability\ Factor$

$10^3\ Life\ Cycles$

Equation 12. $S = 0.75S_u$

Nomenclature:

$S = 0.75S_u = 10^3\ Cycle\ Strength\ For\ Axial\ Loads$

$S_u = Tensile\ Strength$

$2 \times 10^5\ Life\ Cycles$

Force

Equation 13 for initial force is used for ordinary applications involving static loading. The higher the initial tension the less likely the member are to separate. For loads tending to shear the bolts, the higher the initial tension the greater the friction forces resisting the relative motion in shear. The external force works to separate the member. The relative magnitudes of the changes in bolt axial force and clamping force depend on the relative elasticity involved. Equations 13, 14, and 15 are used to calculate initial, axial bolt, and clamping forces, respectively.

Equation 13. $F_i = 0.9A_tS_p$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_y = Yield\ Strength$

Equation 14. $F_b = F_i + \frac{K_b}{K_b+K_c}(F_e)$

Nomenclature:

$F_b = Bolt\ Axial\ Force$

$F_i = Initial\ Tension\ Force$

$F_e = External\ Force$

$K_c = Joint\ Stiffness$

$K_b = Bolt\ Stiffness$

Equation 15. $F_c = F_i + \frac{K_c}{K_b+K_c}(F_e)$

Nomenclature:

$F_c = Clamping\ Force$

$F_i = Initial\ Tension\ Force$

$F_e = External\ Force$

$K_c = Joint\ Stiffness$

$K_b = Bolt\ Stiffness$

Alternating/Mean/Initial Stress

Alternating, mean, and initial stresses are multiplied by the fatigue stress concentration factor and correction is made for yielding and resultant residual stresses if the calculated peak stress exceeds the material yield strength. Alternating stress is produced when forces act alternatively in opposite directions. Mean stress is the average stress per unit area on an object. Initial stress is stress existing in an object that is not subjected to external forces, except gravity. Equations 16, 17, and 18 are used to calculate alternating, mean, and initial stress, respectively.

Equation 16. $\sigma_a = \frac{(F_b)_a}{A_t}\left(K_f\right)$

Nomenclature:

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

$A_t = Tensile\ Stress\ Area$

Equation 17. $\sigma_m = \frac{(F_b)_m}{A_t}\left(K_f\right)$

Nomenclature:

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_m = Mean\ Stress$

$(F_b)_m = Mean\ Bolt\ Axial\ Force$

$A_t = Tensile\ Stress\ Area$

Equation 18. $\sigma_i = \frac{F_i}{A_t}\left(K_f\right)$

Nomenclature:

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$A_t = Tensile\ Stress\ Area$

RESULTS

Bolt Requirements

The tie rod hydraulic cylinder analyzed in this experiment can handled a maximum shock load of 5,000 lb/in^2. The outside diameter of the cylinder end cap was 4.25 in and the inside diameter of the cylinder is 3.5 in. Using the total clamping force equation given in Design of Machine Element's class lecture, a force of 22,825.63 lb was calculated. The minimum number of bolts required, to meet the required clamping force, is two bolts. To meet the spacing requirements, four bolts must be used in the design of the hydraulic cylinder.

Specifications:

$Distance\ Between\ Nonadjacent\ Bolts = 4\ in$

$Pressure = 5{,}000\ lb/in^2$

$D_o = 4.25\ in$

$D_i = 3.5\ in$

Equation 19. $Total\ Clamping\ Force = (pressure)\left(\frac{\pi}{4}\right)\left(D_o^2 - D_i^2\right)$

$$Total\ Clamping\ Force = (5000)\left(\frac{\pi}{4}\right)(4.25^2 - 3.5^2) = 22{,}825.63\ lb$$

Nomenclature:

$D_o = outside\ diameter$

$D_i = inside\ diameter$

Specifications:

$Bolt\ Diameter = 0.5\ in$

$A_t = 0.1419\ in^2$ [1]

$S_p = 12{,}000\ lb/in^2$ [1]

Equation 20. $Damping\ Force\ Per\ Bolt\ @\ 90\%\ \ S_p = (A_t)(0.9)(S_p)$

$Damping\ Force\ Per\ Bolt\ @\ 90\%\ \ S_p = (0.1419\ in^2)(0.9) = \ 15{,}325.2\ lb/in^2$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_p = Proof\ Load\ Strength$

Equation 21. $$\frac{Force}{Bolt} = \frac{Total\ Clamping\ Force}{Damping\ Force\ Per\ Bolt} = \frac{22{,}825.63\ lb}{15{,}325.2\ lb} = 1.49\ \approx 2\ bolts$$

Spacing Requirements

Two Bolts:

Equation 22. $$Spacing = \frac{(Distance\ Between\ Bolts)(\pi)}{\#\ of\ Bolts} = \frac{(4\ in)(\pi)}{2} = 6.28\ in$$

Equation 23. $$Ratio = \frac{Spacing}{Bolt\ Diameter} = \frac{6.28\ in}{0.5\ in} = 12.57$$

Four Bolts:

Equation 24. $$Spacing = \frac{(Distance\ Between\ Bolts)(\pi)}{\#\ of\ Bolts} = \frac{(3\ in)(\pi)}{3} = 3.14\ in$$

Equation 25. $$Ratio = \frac{Spacing}{Bolt\ Diameter} = \frac{3.14\ in}{0.5\ in} = 6.28$$

- Based upon our analysis, only 4 bolts satisfy the requirements for the pressure in the cylinder.

Safety Factor

The specifications sheet for the TX 2500 model of the hydraulic cylinder lists the maximum constant load at 16,560 lb. Assuming an evenly distributed load, the normal load on each bolt is 4,140 lb. Using the design overload and normal load, the safety factor is calculated to be 4.11.

Equation 26. $$Normal\ Load\ Per\ Bolt = \frac{Normal\ Load}{\#\ of\ Bolts} = \frac{16560\ lb}{4} = 4140\ lb$$

Specifications:

$A_t = 0.1419\ in^2$ [1]

$S_p = 120{,}000\ lb/in^2$ [1]

$S_y = 130{,}000\ lb/in^2$ [1]

$S_u = 150{,}000\ lb/in^2$ [1]

$TPI = 13$

Equation27. $SF = \frac{Design\ Overload}{Normal\ Load} = \frac{(S_p)(A_t)}{Normal\ Load} = \frac{(120{,}000\ lb/in^2)(0.1419\ in^2)}{(4140\ lb)} = 4.11$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_p = Proof\ Load\ Strength$

$S_y = Yield\ Strength$

$S_u = Tensile\ Strength$

$TPI = Threads\ Per\ in$

Thread Engagement Requirement

Tensile strength and nut shear strength equations were used to find the thickness of the nut required and the minimum number of threads required to be engaged to ensure failure would not occur. The tensile strength of the nut was calculated to be 17,028 lb. The minimum nut thickness must be 0.6922 in with a minimum of nine threads engaged.

Specifications:

$\mathrm{A_t} = 0.1419\ \mathrm{in}^2$ [1]

$\mathrm{S_p} = 120{,}000\ \mathrm{lb/in}^2$ [1]

Equation 28. $Tensile\ Force = A_t S_y = (0.1419\ in^2)(120{,}000\ lb/in^2) = 17028\ lb$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_y = Yield\ Strength$

Specifications:

$S_y = 36{,}000\ lb/in^2$ [1]

Equation 29. $S_{ys} = 0.58S_y \rightarrow 0.58(36{,}000\ lb/in^2) = 20{,}880\ lb/in^2$

Nomenclature:

$S_{ys} = Shear\ Yield\ Strength$

$S_y = Yield\ Strength$

Specifications:

$S_y = 36{,}000\ lb/in^2$ [1]

$Nut\ Shear\ Strength = 17028\ lb$

Equation 30. $Nut\ Shear\ Force = (\pi)(d)(0.75t)(S_{ys}) \rightarrow (\pi)(d)(0.75t)(0.58)(36000)$

$= 17028\ lb$

$t = 0.6922\ in$

Nomenclature:

$S_{ys} = Shear\ Yield\ Strength$

$S_y = Yield\ Strength$

$t = thickness$

Equation 31. $\#\ of\ Thread\ Engagement = (Thickness)(TPI) = (0.6922)(13)$

$= 9\ threads$

Reversed Axially Loaded

Several different life cycles were calculated to compare the infinite life of the bolts. The Moore [1] fatigue tests were used to determine the fatigue strength characteristics of materials under a standardized and highly restricted set of conditions.

10^6 Life Cycles

Specifications:

$S_n' = 75{,}000\ lb/in^2$

$C_L = 1$ [1]

$C_G = 0.8$ [1]

$C_s = 0.9$ [1]

$C_T = 1$ [1]

$C_R = 1$ [1]

Equation 32. $S_n = S_n^{'} C_L C_G C_s C_T C_R \rightarrow 0.5(150)(1)(0.8)(0.9)(1)(1) = 43{,}200\ lb/in^2$

Nomenclature:

$S_n = 10^6\ Cycle\ Strength\ For\ Fatigue\ Life$

$S_n^{'} = 0.5 S_u$

$S_u = Tensile\ Strength$

$C_L = Load\ Factor$

$C_G = Gradient\ Factor$

$C_s = Surface\ Factor$

$C_T = Temperature\ Factor$

$C_R = Reliability\ Factor$

10^3 Life Cycles

Specifications:

$S_u = 150{,}000\ lb/in^2$

Equation 33. $S = 0.75 S_u = 0.75(150) = 112{,}500\ lb/in^2$

Nomenclature:

$S = 0.75 S_u = 10^3\ Cycle\ Strength\ For\ Axial\ Loads$

$S_u = Tensile\ Strength$

2×10^5 Life Cycles – $58{,}000\ lb/in^2$

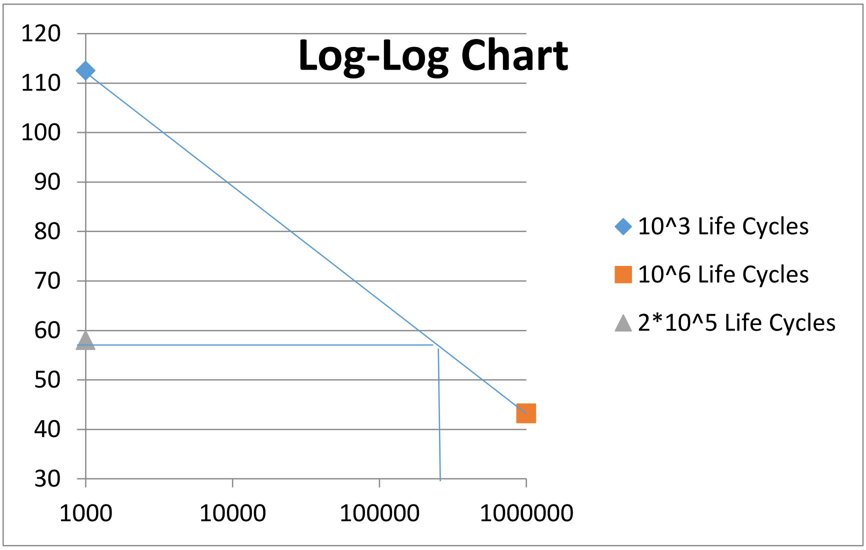

Figure 6. Solving for $2 * 10^5$ Life Cycles

Maximum Value of P for Infinite Life (0 – P)

The initial force was found to be 15,330 lb/in^2. The equation used to find the initial force can be used for ordinary applications involving static loading. The maximum and minimum bolt axial force was used to find the alternating and mean bolt axial force. The alternating and mean bolt axial force was used to find equations for the mean and alternating stresses. 10^6 Life cycle fatigue strength was calculated and graphed to assist in finding a value for alternating stress. Using the value of alternating stress, the maximum value of P that would not cause eventual fatigue failure of the bolts was calculated to be 22,704 lb. A safety factor with respect to eventual bolt fatigue failure was found using the values for alternating and initial stress. The safety factor was calculated to be 2.

Specifications:

$$A_t = 0.1419\ in^2 \quad [1]$$

$$S_p = 120{,}000\ lb/in^2 \quad [1]$$

Equation 34. $F_i = 0.9 A_t S_p \rightarrow (0.9)(0.1419\ in^2)(120{,}000\ lb/in^2) = 15{,}330\ lb$

Nomenclature:

$A_t = Tensile\ Stress\ Area$

$S_y = Yield\ Strength$

Equation 35. $F_b = F_i + \frac{K_b}{K_b+K_c}(F_e)$

Nomenclature:

$F_b = Bolt\ Axial\ Force$

$F_i = Initial\ Tension\ Force$

$F_e = External\ Force$

$K_c = Joint\ Stiffness$

$K_b = Bolt\ Stiffness$

Specifications:

$F_i = 15{,}330\ lb\ (i.e.\ 15.33\ kips)$

$K_c = 4K_b$

Equation 36. $(F_b)_{max} = F_i + \frac{K_b}{K_b+K_c}(F_e) \rightarrow 15.33 + \left(\frac{1}{4}\right)\left(\frac{P}{4}\right)$

Nomenclature:

$F_b = Bolt\ Axial\ Force$

$F_i = Initial\ Tension\ Force$

$F_e = External\ Force$

$K_c = Joint\ Stiffness$

$K_b = Bolt\ Stiffness$

$P = External\ Load$

Specifications:

$F_i = 15.33\ kips$

$K_c = 4K_b$

Equation 37. $(F_b)_{min} = F_i + \frac{K_b}{K_b+K_c}(F_e) \rightarrow 15.33 + \left(\frac{1}{4}\right)(0) = 15.33\ kips$

Nomenclature:

$F_b = Bolt\ Axial\ Force$

$F_i = Initial\ Tension\ Force$

$F_e = External\ Force$

$K_c = Joint\ Stiffness$

$K_b = Bolt\ Stiffness$

$P = External\ Load$

Specifications:

$(F_b)_{max} = 15.33 + \left(\frac{1}{4}\right)\left(\frac{P}{4}\right)$

$(F_b)_{min} = 15.33$

Equation 38. $$(F_b)_a = \frac{(F_b)_{max} - (F_b)_{min}}{2} = \frac{15.33 + \frac{P}{16} - 15.33}{2} = \frac{P}{32}$$

Nomenclature:

$(F_b)_{max} = Maximum\ Bolt\ Axial\ Force$

$(F_b)_{min} = Minimum\ Bolt\ Axial\ Force$

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

Specifications:

$(F_b)_{max} = 15.33 + \left(\frac{1}{4}\right)\left(\frac{P}{4}\right)$

$(F_b)_{min} = 15.33$

Equation 39. $$(F_b)_m = \frac{(F_b)_{max} + (F_b)_{min}}{2} = \frac{15.33 + \frac{P}{16} + 15.33}{2} = 15.33 + \frac{P}{32}$$

Nomenclature:

$(F_b)_{max} = Maximum\ Bolt\ Axial\ Force$

$(F_b)_{min} = Minimum\ Bolt\ Axial\ Force$

$(F_b)_m = Mean\ Bolt\ Axial\ Force$

10^6 Life Cycles

Specifications:

$S_n' = 75{,}000\ lb/in^2$

$C_L = 1$ [1]

$C_G = 0.8$ [1]

$C_s = 0.9$ [1]

$C_T = 1$ [1]

$C_R = 1$ [1]

Equation 40. $S_n = S_n' C_L C_G C_s C_T C_R \rightarrow 0.5(150)(1)(0.9)(1)(1)(1) = 67{,}500\ lb/in^2$

Nomenclature:

$S_n = 10^6\ Cycle\ Strength\ For\ Fatigue\ Life$

$S_n' = 0.5 S_u = 10^6\ Cycle\ Strength\ For\ Axial\ Loads$

$S_u = Tensile\ Strength$

$C_L = Load\ Factor$

$C_G = Gradient\ Factor$

$C_s = Surface\ Factor$

$C_T = Temperature\ Factor$

$C_R = Reliability\ Factor$

Specifications:

$K_f = 3.0$ [1]

$(F_b)_a = \frac{P}{32}$

$A_t = 0.1419\ in^2$ [1]

Equation 41. $\sigma_a = \frac{(F_b)_a}{A_t}(K_f) \rightarrow \frac{\frac{P}{32}}{0.1419}(3)$

Nomenclature:

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

$A_t = Tensile\ Stress\ Area$

Specifications:

$K_f = 3.0$ [1]

$(F_b)_m = \frac{P}{32} + 15.33$

$A_t = 0.1419\ in^2$ [1]

Equation 42. $\sigma_m = \frac{(F_b)_m}{A_t}(K_f) \rightarrow \frac{\frac{P}{32}+15.33}{0.1419}(3)$

Nomenclature:

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_m = Mean\ Stress$

$(F_b)_m = Mean\ Bolt\ Axial\ Force$

$A_t = Tensile\ Stress\ Area$

Specifications:

$K_f = 3.0$ [1]

$F_i = 15{,}330\ lb/in^2$

$A_t = 0.1419\ in^2$ [1]

Equation 43. $\sigma_i = \frac{F_i}{A_t}(K_f) \rightarrow \frac{15.33}{0.1419}(3) = 324{,}100\ lb/in^2$

- Based upon our analysis, the bolt will yield first.

Nomenclature:

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$A_t = Tensile\ Stress\ Area$

Application Line

Specifications:

$\sigma_a = 15{,}000\ lb/in^2$

$S_y = 130{,}000\ lb/in^2$ [1]

Equation 44. $(F_b)_{max} = F_i + F_a \rightarrow \frac{(F_b)_{max}}{A_t}(K_f) = \frac{F_i}{A_t}(K_f) + \frac{F_a}{A_t}(K_f)$

$\rightarrow S_y = \sigma_i + \sigma_a$

$130 = \sigma_i + 15$

$\sigma_i = 115{,}000\ lb/in^2$

Nomenclature:

$(F_b)_{max} = Maximum\ Bolt\ Axial\ Force$

$F_a = Alternating\ Force$

$F_i = Initial\ Force$

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$A_t = Tensile\ Stress\ Area$

$S_y = Yield\ Strength$

Specifications:

$K_f = 3.0$ [1]

$(F_b)_a = \frac{P}{32}$

$A_t = 0.1419\ in^2$ [1]

$\sigma_a = 15{,}000\ lb/in^2$

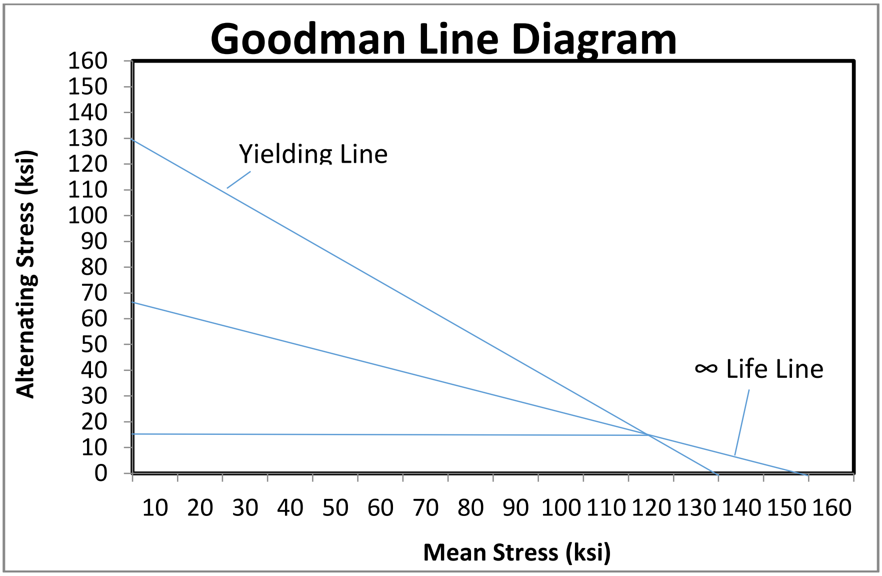

Figure 7. Goodman Line Diagram

Equation 45. $\sigma_a = \frac{(F_b)_a}{A_t}(K_f) \rightarrow 15 = \frac{\frac{P}{32}}{0.1419}(3) \rightarrow P = 22{,}704\ lb$

Nomenclature:

$(F_b)_a = Alternating\ Bolt\ Axial\ Force$

$K_f = Fatigue\ Stress\ Concentration\ Factor$

$\sigma_a = Alternating\ Stress$

$A_t = Tensile\ Stress\ Area$

Safety Factor With Respect To Eventual Bolt Fatigue Failure

Specifications:

$\sigma_a = 15{,}000\ lb/in^2$

$\sigma_i = 7{,}500\ lb/in^2$

Equation 46. $SF = \frac{Design\ Overload}{Normal\ Load} = \frac{\sigma_a}{\sigma_i} = \frac{15}{7.5} = 2$

Nomenclature:

$\sigma_a = Alternating\ Stress$

$\sigma_i = Initial\ Stress$

LIMITATIONS

The assumptions of the material used for the bolt and nut restricted the accuracy of the calculations.

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